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**UNIVERSITY ROLE IN ASTRONAUT
LIFE SUPPORT SYSTEMS: PORTABLE
THERMAL CONTROL SYSTEMS**

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for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • NOVEMBER 1971



0061026

1. Report No. NASA CR-1852		2. Government Accession No.		3. Recipient's Catalog No.	
4. Title and Subtitle UNIVERSITY ROLE IN ASTRONAUT LIFE SUPPORT SYSTEMS: PORTABLE THERMAL CONTROL SYSTEMS				5. Report Date November 1971	
				6. Performing Organization Code	
7. Author(s) ARYE R. EPHRATH				8. Performing Organization Report No.	
				10. Work Unit No. 127-53-27-22-10	
9. Performing Organization Name and Address DEPARTMENT OF AERONAUTICS AND ASTRONAUTICS SCHOOL OF ENGINEERING, MAN-VEHICLE LAB MASSACHUSETTS INSTITUTE OF TECHNOLOGY CAMBRIDGE, MASSACHUSETTS 02139				11. Contract or Grant No. NGR-22-009-312	
				13. Type of Report and Period Covered CONTRACTOR REPORT	
12. Sponsoring Agency Name and Address NATIONAL AERONAUTICS & SPACE ADMINISTRATION WASHINGTON, D. C. 20546				14. Sponsoring Agency Code	
15. Supplementary Notes This program is supported by Bioenvironmental Systems, Life Science Program, Office of Manned Space Flight, to provide improved technical requirements communication with the University scientific and engineering community.					
16. Abstract ONE OF THE MOST VITAL LIFE-SUPPORT SYSTEMS IS THAT USED TO PROVIDE THE ASTRONAUT WITH AN ADEQUATE THERMAL ENVIRONMENT. THIS PAPER REVIEWS STATE-OF-THE-ART TECHNIQUES FOR COLLECTING AND REJECTING EXCESS HEAT-LOADS FROM THE SUITED ASTRONAUT. EMPHASIS IS PLACED ON PROBLEM AREAS WHICH EXIST AND WHICH MAY BE SUITABLE TOPICS FOR UNIVERSITY-TYPE RESEARCH. AREAS COVERED INCLUDE THERMAL-CONTROL REQUIREMENTS AND RESTRICTIONS, METHODS OF HEAT ABSORPTION AND OF HEAT REJECTION OR STORAGE, AND COMPARISON BETWEEN EXISTING METHODS AND POSSIBLE FUTURE TECHNIQUES. AN ATTEMPT IS MADE TO STIMULATE NEW IDEAS BASED ON PRESENT KNOWLEDGE.					
17. Key Words (Suggested by Author(s)) THERMAL CONTROL HEAT REJECTION HEAT STORAGE				18. Distribution Statement UNCLASSIFIED - UNLIMITED	
19. Security Classif. (of this report) UNCLASSIFIED		20. Security Classif. (of this page) UNCLASSIFIED		21. No. of Pages 65	
				22. Price* \$3.00	

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FOREWORD

The Life Sciences Program Office of the National Aeronautics and Space Administration's Office of Manned Space Flight is vitally interested in promoting and developing new ideas which will advance the technology in thermal control systems for portable life support systems. Many excellent concepts and operating subsystems for crew equipment technology have been translated into successful hardware demonstration in NASA laboratories and by associated contractors. The involvement of academic laboratories and personnel has not been as great as was originally hoped. Perhaps this has been due to the fact that academic investigators were not aware of the critical problem areas in portable life support systems nor were they cognizant of the fact that NASA is interested in joining with colleges and universities to develop new ideas to solve future manned space flight problems.

This report on Portable Thermal Control Systems is intended to introduce the reader to some of the existing Crew Equipment Technology involved in maintaining astronauts during extravehicular activities in support of mission requirements and to pinpoint areas where problems exist. We encourage you to study this report. If, in your research, you have already developed new ideas, theories, chemicals, etc., which would be applicable to NASA's problems, we hope you will feel inclined to contact us to see whether a joint research effort can be initiated.

Walton L. Jones
Deputy NASA Director for Life Sciences

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I. INTRODUCTION

The most difficult problem encountered in the design of space-oriented life-support systems is that of providing the astronaut with an adequate thermal environment at all times^{24,25}. While it is true that man can adapt satisfactorily to a fairly wide range of environmental conditions⁸ he requires nonetheless that heat removal from the body be accomplished within the narrow margins of comfortable temperature level^{8,34}. The problem is complicated further by the following limitations imposed on the system:

Widely varying heat inputs. The inputs into the cooling system are the astronaut's metabolic heat production, heat generated by subsystems within the portable life-support system and external heat inputs from the environment, such as thermal radiation from the sun, planetary surfaces, and the spacecraft. These external heat inputs vary considerably. Furthermore, the added constraints of a pressurized soft space suit, low or zero gravity, and low-traction environments result in a wide range of metabolic rates; these vary by about one order of magnitude. Thus the cooling system requires the capability to adapt quickly to a wide range of heat-removal rates.

Limited heat-transfer modes. The heat-transfer modes which are available for thermal regulation of the body in the terrestrial environment are almost completely eliminated in the space suit environment. Radiative heat-transfer may be essentially eliminated for an astronaut dressed in a thermally insulated suit. Natural convective heat transfer mode loses its effectiveness at reduced gravity and at the lower ambient pressure within the space suit. Latent heat of evaporation, one of the most important thermoregulatory mechanisms, is seriously impaired as sweating must be severely restricted within the suit for reasons of comfort and contaminants control. The cooling system must rely, therefore

on other methods of heat transport and heat rejection.

These are some of the severest limitations imposed on the cooling system by the peculiarities of the space environment. These limitations are in addition to such usual restrictions on space hardware as limited weight, bulk and power requirements and the demand for maximum reliability.

Numerous schemes have been proposed to answer the problem of cooling a man in space. Most of these methods require further improvements, optimization and refinements that research on the university level may be able to provide. This report examines a few of the more promising methods proposed to date and points out the areas in which more work needs to be done.

1.1 System Requirements

The heat-balance for the human body is given by

$$\dot{Q}_{\text{met}} = \dot{Q}_w + \dot{Q}_e \pm \dot{Q}_s \pm \dot{Q}_c \pm \dot{Q}_R \pm \dot{Q}_{\text{storage}} \quad (1-1)$$

where \dot{Q}_{met} = metabolic heat production rate
 \dot{Q}_w = external work rate
 \dot{Q}_e = heat transfer rate due to the evaporation of sweat from the body surface
 \dot{Q}_s = net sensible heat transfer rate due to convection and respiration
 \dot{Q}_c = heat transfer rate due to conduction
 \dot{Q}_R = net radiative heat transfer rate

In the space environment \dot{Q}_e , \dot{Q}_s , and \dot{Q}_c end up as inputs into the cooling system and may be designated as $\dot{Q}_{\text{thermal control}}$

$$\dot{Q}_{\text{thermal control}} = \dot{Q}_{\text{met}} - \dot{Q}_w \pm \dot{Q}_R \pm \dot{Q}_{\text{storage}} \quad (1-2)$$

While small heat storage changes can be tolerated for limited periods of time, continuous steady-state operation requires that \dot{Q}_{storage} must be zero. Thus,

$$\dot{Q}_{\text{thermal control}} = \dot{Q}_{\text{met}} - \dot{Q}_w \pm \dot{Q}_R \quad (1-3)$$

The "plus" sign on \dot{Q}_R applies in case of net heat transfer from space into the suit.

It has been found⁸ that the metabolic cost increases very significantly when work is performed in a pressurized soft non-constant volume joint space suit, and \dot{Q}_w is negligible compared to \dot{Q}_{met} ; furthermore, a large part of the increased energy output is expended on moving the suit, and therefore, the heat generated remains in the system. Consequently, no allowance should be made for work output in formulating a conservative estimate of the required heat-removal capability of the cooling system:

$$\dot{Q}_{\text{thermal control}} = \dot{Q}_{\text{met}} \pm \dot{Q}_R \quad (1-4)$$

Metabolic Rate

Metabolic heat-generation rate depends on many factors such as body mass, body surface area, deep body temperature, restriction of clothing and, of course, type of activity. Table 1-1 shows a number of typical metabolic rates for various activities.

A soft non-constant volume joint space suit pressurized to 3.5 psi will increase the metabolic rate in some cases by 178-367 percent relative to the same activity in normal clothing⁸. Table 1-2 shows the values of metabolic rates obtained while performing treadmill exercise in a 1-g and simulated 1/6-g environments²⁷, with the suit pressurized to 3.5 psi above ambient.

TABLE 1-1:
TYPICAL METABOLIC HEAT EXPENDITURES^{8,25}

Activity	Metabolic Rate, Btu/hr
Sleeping man	300
sitting at rest	400
standing relaxed	500
Instrument landing, DC-4	590
Slow walking	900
Walking at 2.5 mph on flat surface	1000
Walking at 3.75 mph	1300
Swimming	2200
Very severe exercise	2600
EVA (Gemini 9)	3000

TABLE 1-2:
METABOLIC RATE - WALKING IN A-5-L SPACE SUIT²⁷

g level	Suit Mode	Metabolic Rate, Btu/hr		
		Resting	1 mph	4 mph
1 - g	Ventilated	348.2±25.4	767±117	----
	Pressurized	----	1164±164	----
1/6-g on inclined plane	Ventilated	326.9±44.8	412±54	739±89
	Pressurized	----	558±112	1346±317
1/6-g on 6 DOF* gimbal	Ventilated	279.0±40.0	401±90	487±65
	Pressurized	----	695±195	737±122

* Degrees of Freedom

Based on these results, as well as on past experience with EVA, a conservative estimate for the metabolic heat rate points towards \dot{Q}_{man} varying between 300-2500 Btu/hr with occasional "spikes" reaching 3500 Btu/hr for short periods of time.

Radiative Heat Transfer

The net radiative heat transfer between a space suit and its environment depends, naturally, on the nature of the thermal environment. The heat exchange taking place among an astronaut in earth-orbit, his spacecraft and deep space depends, among other factors, on the radius of the orbit and its orientation (see Section 4.2) and will, of course, be quite different from the radiative heat-exchange for the same astronaut on the lunar surface or inside a lunar crater. In general, the net radiative heat exchange is a function of the surface area of the space suit, its thermal characteristics (such as surface emittance and absorptance), the fourth power of the absolute effective temperatures of the suit surface and the external heat source and the orientation of the astronaut with respect to this source. An "equivalent network" analysis can be performed³⁸ to determine the net heat transfer between two or more bodies. As a conservative figure which is applicable to a wide range of conditions a value of \dot{Q}_R varying between +250 Btu/hr and -350 Btu/hr is accepted as a safe requirement imposed on the thermal control system^{20,28} (see also Section 3.3 of this Report).

Additional Partition of Thermal Loads

$\dot{Q}_{\text{metabolic}}$ was found above to range from 300 to 2500 Btu/hr, with average value of 2000 Btu/hr and short-duration "spikes" reaching 3500 Btu/hr. $\dot{Q}_{\text{radiation}}$ was taken as ranging from 250 Btu/hr input into the space suit from external heat sources to 350 Btu/hr heat loss

from the suit. In addition, the space suit system contains various equipment such as communications and accessories (circulation pump, fan etc.). The heat output of these sources was estimated²⁸ not to exceed 125 Btu/hr. The total capacity of the thermal regulatory system should then be 2875 Btu/hr, as shown in Figure 1-1.

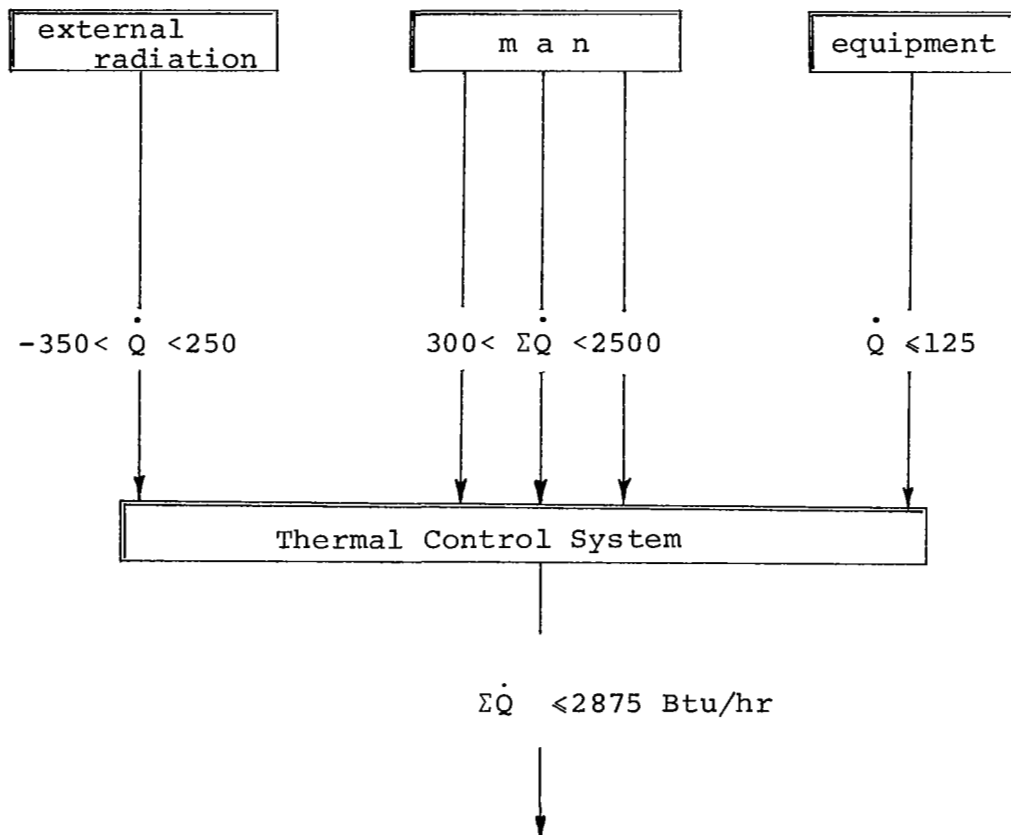


Figure 1-1: Thermal Control Requirements

In general, the following requirements are imposed²⁰ on the thermal control subsystem:

	Space <u>EVA</u>	Lunar <u>EVA</u>
Mission duration (maximum), hr	4	5
Emergency operation, hr	0.5	2
Average metabolic rate, Btu/hr	2000	1600
Sustained maximum metabolic rate, Btu/hr	2500	2500
Metabolic spikes, Btu/hr	3500	--
Spikes duration, min	10	--
Emergency average MR, Btu/hr	2000	1600
Emergency maximum MR, Btu/hr	2500	2500
Maximum external heat loads:		
Input, Btu/hr	+250	+250
Loss, Btu/hr	-350	--
Total mission load capability, Btu	9000	11200

In addition, the system is subjected to the general requirements imposed on all space hardware, such as:

1. The system must be reliable.
2. The system has to be compatible with the space suit.
3. Capability for recharging in space is required.

1.2 Restrictions

The thermal control system must fulfil the requirements, as stated in Section 1.1, while subjected to certain constraints. These restrictions stem from considerations of both the astronaut's comfort and safety and the space-vehicle's limitations. The system therefore has to remove heat from the astronaut's body at acceptable temperature levels and at the same time conform to the demand for minimum bulk, weight and power requirements.

Comfort

Several comfort indices have been proposed to date, such as the ASHRAE Comfort Chart, operative temperature or the PS4R index^{8,29}. Most of the work done in this field, however, has been air-crew oriented and does not apply directly to space-suit or reduced-gravity environments.

The Evaporative Capacity Comfort Criterion⁸, first developed by C.E.A. Winslow in 1937, relies on the body's thermoregulatory mechanism in using the sweat-rate, converted to percentage of maximum evaporative capability under the given conditions, as the comfort index. This index, therefore, is directly applicable to all environments. Naturally, the maximum evaporative capability depends on the environmental parameters and the nature of heat transfer modes which are available. Figure 1-2, for example, shows the relation between mean skin temperature and metabolic rate when the primary mode of heat transfer is by conduction (as by a liquid cooled garment).

Mean skin temperature is, as the name implies, the local skin temperature averaged over the body surface area. Local skin temperatures vary greatly among different body regions,

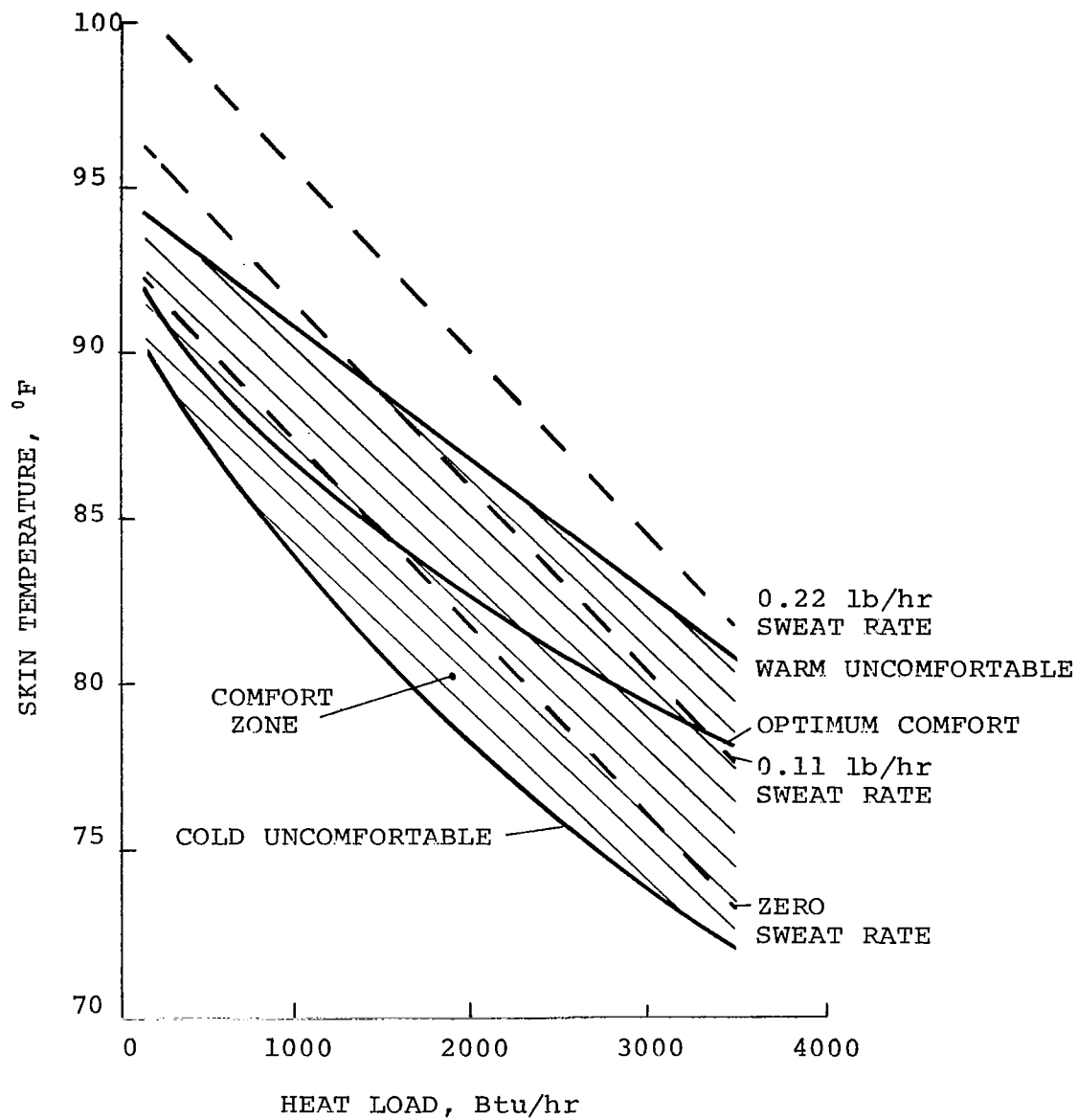


Fig. 1-2. CONDUCTION COOLING COMFORT¹⁰

as Table 1-3 illustrates.

TABLE 1-3:
SKIN TEMPERATURE AND HEAT LOSS DISTRIBUTIONS
FOR THERMALLY COMFORTABLE MAN AT REST⁹

Region	Skin Temperature, °F	Fraction of Heat Loss, %	Fraction of Heat Loss per Unit Area, %/m ²
Head	94.08	3.74	28.3
Chest	94.08	7.66	
Abdomen	94.08	4.20	
Back	94.08	11.60	45.9
Buttocks	94.08	7.76	
Thighs	91.20	11.20	
Calves	87.24	13.64	} 42.3
Feet	83.28	9.35	72.1
Arms	91.20	7.85	
Forearms	87.24	8.04	} 62.6
Hands	83.23	14.96	158.5
Mean	91.20	Total: 100.00	

(It should be noted that this distribution of skin temperatures may change significantly when thermal stresses are present¹.)

The constraint on acceptable skin temperature must, therefore, include at least three parameters: Mean skin temperature as required for comfort, plus maximum and minimum values for local skin temperatures; these are usually set at 104°F (40°C) and 68°F (20°C), respectively³.

Other Considerations

In addition to the demand for a comfortable thermal environment for the EV astronaut, the crew's safety and the space vehicle limitations must be taken into account. This imposes on the thermal control system additional requirements, such as:

1. Total weight, bulk and power requirements should be minimized.
2. System components must be stowed aboard the spacecraft when not in use.
3. Weight and volume of expendable and non-reusable components should be kept to a minimum.
4. Toxic and combustible materials should be avoided.
5. When in operation the interference of the thermal control system with the astronaut's activities should be minimal.

1.3 Configuration

From the foregoing discussion it seems obvious that the thermal control system should perform four different tasks:

1. Decouple the astronaut from the external thermal environment.
2. Absorb excess heat from the astronaut's body.
3. Transport the absorbed heat away from the body.
4. Reject the heat to a heat-sink.

Most of the thermal control schemes which have been proposed to date consist of a heat-transport subsystem which absorbs the heat generated by the astronaut and transports it to a heat exchanger, and another subsystem which rejects the heat to an integral heat sink or to space. The function of decoupling the astronaut from his thermal environment is assumed by the thermal insulation layers built into the suit.

Heat collection and transport schemes are described in Chapter II of this Report, and heat rejection systems in Chapter III. Also, a number of thermal control schemes have been proposed in which the same subsystem performs both functions, absorbing excess heat from the astronaut's body and rejecting it; these are described in Chapter IV as integrated systems.

It should be noted that it generally is very difficult, if not impossible, to devise "absolutely optimal" systems, and thermal control systems are no exception to this rule. An important set of parameters in optimization is mission type and mission profile, and a system which looks very promising for one class of EVA missions may be totally inadequate for another. This is why promising concepts of environmental control schemes are described in this Report with their respective advantages and disadvantages. When selecting a system for a particular mission it is the trade-off between these advantages and disadvantages that should be taken into consideration.

1.4 Recommendations for University Research

- a. Better understanding of human thermal processes and heat balance as applicable to space is needed.
- b. Better prediction capability of metabolic rates as associated with EVA at reduced gravity should be developed, both for soft, hybrid, and hard suits.
- c. Determine acceptable skin temperatures distributions and deep-body heat storage limits in space environments.
- d. Develop comfort criteria applicable to EVA under different heat-transfer modes.

II. HEAT TRANSPORT SYSTEMS

Several different methods have been developed to absorb heat from the astronaut's body and transport it to a heat exchanger. They employ the whole spectrum of available heat transfer modes. Some were developed for soft space suits, some with the constant-volume "hard shell" concept in mind, and some are compatible with both types.

2.1 Gas Ventilation Loop

Ventilated suits were originally developed for aircrews operating conventional aircraft. The thermal loads which are encountered under such operating conditions are substantially lower than the loads present during EVA. Indeed, the Gemini 9 EVA had to be aborted because of insufficient cooling capability of the space suit.

Nevertheless, ventilation cooling must be considered. A space suit must be ventilated, quite apart from temperature control consideration, both because of contaminant control requirements and in order to remove sensible and insensible water from the suit^{19,21}.

Forced-convection heat transfer rate is given by²⁹

$$\dot{Q}_c = h_c A (T_s - T_{vg}) \quad (2-1)$$

where h_c = convective heat-transfer coefficient; this, in general, is a function of the ventilating gas flow rate, thermal properties of the gas and body geometry.

A = surface area
 T_{vg} = ventilating gas inlet temperature
 T_s = mean skin temperature

In addition, if the H_2O vapor pressure in the ventilating gas is less than the saturation pressure at skin temperature there will be evaporative mode of heat transfer present, due to the evaporation of sweat. The rate of evaporative heat transfer is given by²⁹

$$\dot{Q}_e = \dot{W}_{H_2O} \cdot h_{fg} \quad (2-2)$$

where \dot{W}_{H_2O} is the rate of water (sweat) evaporation and h_{fg} is the enthalpy of vaporization at skin temperature, ≈ 1050 Btu/lb. \dot{W}_{H_2O} is a function of vapor diffusivity, vapor pressure at the skin surface and in the ventilating gas, total pressure of the ventilating gas, its temperature and the thickness of the boundary layer^{11, 29}. The combined effect of convection and evaporation can be reduced to a rather simple expression when the following assumptions are made:

1. The ventilating gas is dry air at 3.5 psia.
2. The astronaut is thermally comfortable.
3. Only small amounts of sweating are present (less than 0.22 lbs/hr).

Under these conditions the ventilation heat transfer rate is given by

$$\dot{Q}_v = 0.24 \times 60 (T_{s,c} - T_{vg}) \dot{W}_v \quad (2-3)$$

where \dot{Q}_v = heat removal rate, Btu/hr

$T_{s,c}$ = skin temperature required for comfort, °F

T_{vg} = ventilating gas inlet temperature, °F

\dot{W}_v = ventilating gas mass flow rate, lb/min

Configuration

Past space suits (MERCURY and GEMINI) relied solely on ventilation by a circulating gas as a means of heat removal from the astronaut's body. In these suits the gas enters through a self-sealing inlet connection in the suit's torso section. A manifold system ducts the gas to the boots, gloves and helmet. The gas then flows back over the legs, arms and torso and removes metabolic heat (see Figure 2-1). A portion of the ventilating gas passes through an integral neck ring and is directed across the visor to reduce fogging and to provide breathing oxygen and remove CO₂. The resulting gas mixture of warm oxygen, carbon dioxide and water vapor is ducted out of the suit at a torso outlet fitting.

Basically, this pattern of ventilating gas flow has not been changed in more advanced suits which depend on other means of primary heat removal.

In tests performed^{8,28} with ventilated GEMINI and early APOLLO suits it was found that the rate of heat removal, consistent with gas temperatures and flow rates acceptable from the standpoint of comfort, could not exceed 750-900 Btu/hr. Figure 2-2, for instance, shows the rate of heat removal of an early APOLLO gas-cooled pressure suit at 3.5 psi and gas flow rate of 15 cfm, versus metabolic rate.

It was this inadequacy of the gas-loop cooling technique, combined with its high specific power requirements, that led to the concept of liquid-loop cooling as the primary method of metabolic-heat collection, in conjunction with a gas loop for humidity and contaminant control. At present, the system requirement is a collection ratio of approximately 80/20^{15,21}.

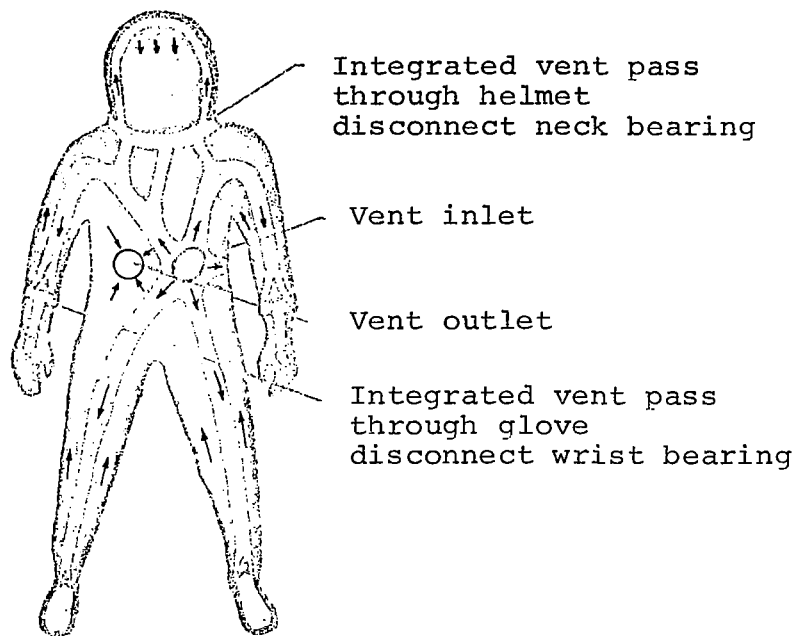


Figure 2-1: Ventilation Distribution System

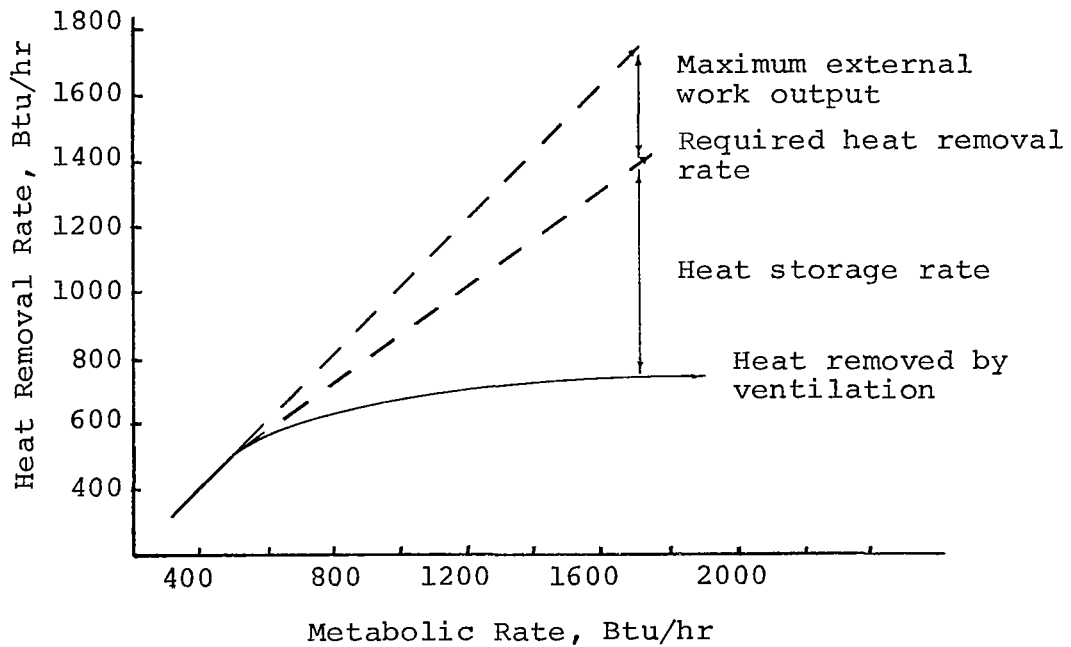


Figure 2-2: Thermal Inadequacy of Ventilated Suits²⁸

2.2 Liquid Loop

Power requirements for heat transfer may be obtained approximately⁹ from

$$\frac{\text{Heat transferred}}{\text{Power required}} = 2500 \frac{\Delta T}{\Delta P} \rho C_p \quad (2-4)$$

where ΔT = temperature gradient, °C

ΔP = pressure gradient, psf

ρ = coolant density, lb/ft³

C_p = coolant heat capacity, Btu/lb°F

As $\rho C_p]_{H_2O} \approx 62 \text{ Btu/ft}^3\text{°F}$, $\rho C_p]_{O_2} \approx 0.0045 \text{ Btu/ft}^3\text{°F}$, it is clear that the power requirement for a given heat removal rate at given temperature and pressure gradients is by three or four orders of magnitude larger for an oxygen-loop system than the power requirement for a comparable water-loop system (water is the most likely heat-transport fluid because of its good heat transport properties, its availability and lack of toxicity⁸). An additional advantage that liquid cooling has over gas cooling is the suppression of thermal sweating and thereby the elimination of dehydration effects¹⁶.

Water-cooled suit was first developed in Farnborough, England, by the British Royal Aircraft Establishment⁹ for use by air crews. It consisted of 40 vinyl tubes (OD 1/8 in., ID 1/16 in.) each 4-6 feet long. The tubes were attached to an undergarment which maintained them in contact with the skin. Design factors of this first suit and a number of subsequent developments are given in Table 2-1.

TABLE 2-1:
LIQUID-COOLED GARMENT DESIGN FACTORS¹⁶

Model	Total tube length, ft	Number of tubes	Garment ^a	Tube pattern ^b	Water inlet ^c
CG1	232	40	CN	FN	EX
CG2	267	40	CN	M	EX
CG3A	300	48	CN	M	EX
CG10	300	48	E	ST	W

^a CN = cotton net; E = elastic

^b FN = follows net; M = meanders; ST = straight

^c EX = extremities (wrists & ankles); W = waist
return outlet has always been at waist

The performance equation of the water-cooled suit was given⁹ as

$$\dot{Q} = \dot{m} C_p \epsilon (33 - T_i) \quad (2-5)$$

$$\epsilon = 1 - \exp(-AU/\dot{m}C_p)$$

where \dot{Q} = cooling rate, Kcal/hr

\dot{m} = coolant flow rate, Kg/hr

C_p = coolant specific heat, cal/gm⁰C

T_i = coolant inlet temperature, ⁰C

ϵ = suit effectiveness

A = total wetted area of the suit, cm²

U = overall heat transfer coefficient of the tubing,
Kcal/cm²hr ⁰C

The equation was based on the assumption of constant skin temperature of 33⁰C. This assumption, however, has since been shown

in a later study to be unjustified³⁴; mean skin temperature was shown to decrease with increasing metabolic rate if the subject is to be thermally neutral (comfortable and perspiring at less than 100 gm/hr; see also Figure 1-2). This, of course, was to be expected if the human body is assumed to offer a constant thermal resistance⁸.

It was also shown³⁴ that the heat-removal rate of a liquid-cooled garment is primarily a function of the inlet coolant temperature and is relatively independent of the coolant flow rate. Consequently, LCGs offer an attractive method of heat collection, combining efficiency with ease of controlling the heat removal rate.

Although the LCG model in which the coolant-filled tubes are in direct contact with the skin seems to be desirable, different schemes of liquid-cooling have been reported in the past^{8,26,32}. These schemes rely on convective heat transfer by a circulating gas or on radiative heat transfer for transport between the body surface and the coolant, which circulates in tubes attached to the interior of the space-suit wall. Heat transfer takes place through a large percentage of the total tube-wall area, as opposed to the small portion of the tubes' surface area that can come in direct contact with the skin in the conductive heat transfer method. It should be noted, however, that while removing the cold tubes from the astronaut's skin may have some merits, from the standpoint of thermal efficiency alone the conduction technique is superior by far to either convection or radiation.

While a liquid loop requires less power than a ventilating-gas loop and liquid coolants in general (and water in particular) have high specific heats, this technique of heat transport is not free of disadvantages. Liquid coolant must be kept from freezing, thus limiting the available temperature gradients in the heat exchanger. Additives which lower the coolant's

freezing point are available, but some of them are toxic; Also, they lower the coolant's specific heat as well. Precautions must be taken to prevent aeration of the liquid coolant, or cavitation and loss of efficiency in the circulation-pump may result.

2.3 Heat Pipes

Gas ventilation and liquid-loop cooling, discussed previously, share a common disadvantage: both require auxiliary equipment such as manifolds, connectors, fan-and-motor or pump and, above all, a power source, all of which add to the weight and complexity of the life-support system. An attempt to overcome this disadvantage was made with the development of the space-suit applicable heat-pipe.

A heat-pipe is a device which transports heat by means of a closed evaporating-condensing cycle. Schematically it can be conceived as a hollow, hermetically-sealed container (see Figure 2-3) the inner wall of which is lined with a capillary wick which, in turn, is soaked with some suitable liquid (the working fluid).

Vapor is produced at the evaporator end. The vapor flows to the condenser end where it condenses, giving up the heat of vaporization. The liquid then returns to the evaporator end through capillary action along the wick lining.

The rate of heat flow in the heat-pipe -- its "conductivity" -- can be controlled if a modification is incorporated in the device, as shown in Figure 2-4. This results in what essentially is a "variable conductance" capability.

The integration of heat pipes into a space suit system as a means of heat absorption and transport from the astronaut's

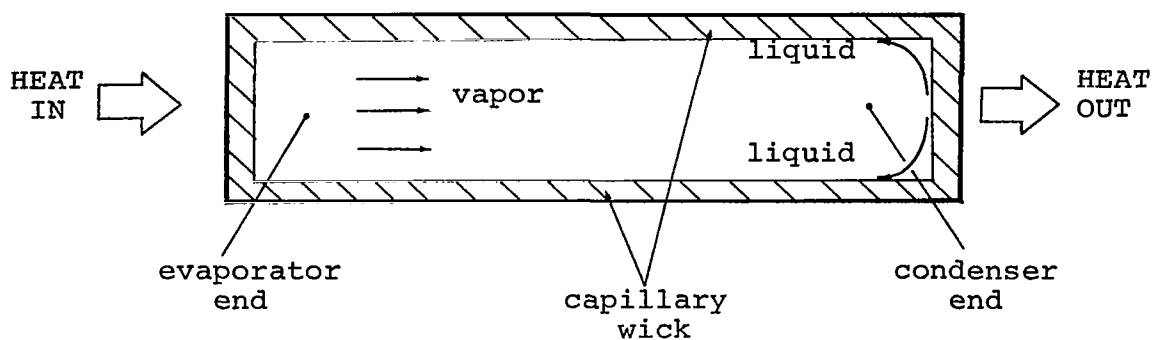


Figure 2-3: Heat Pipe

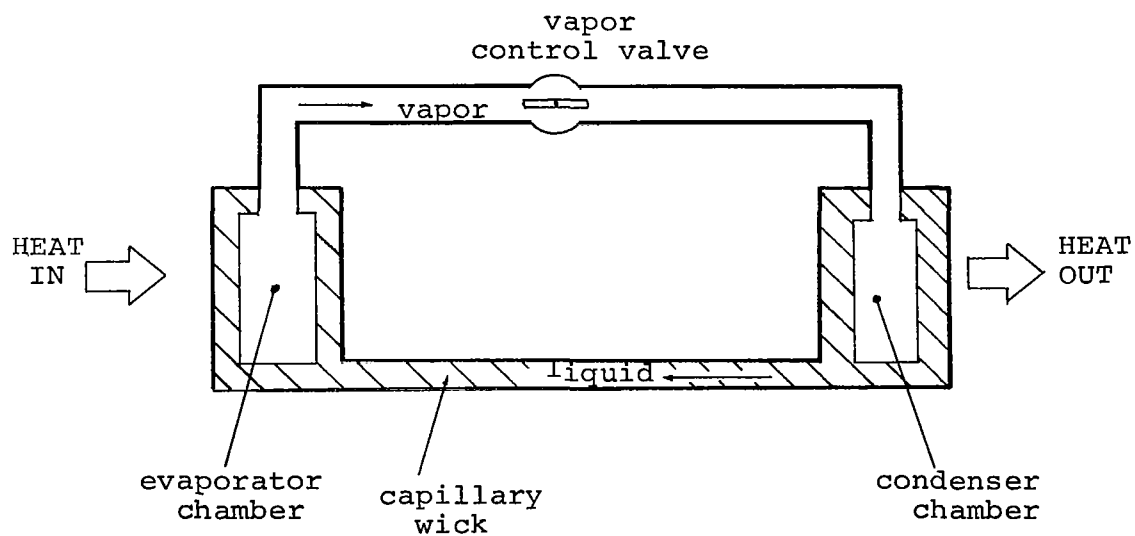


Figure 2-4: Variable Conductance
Heat Pipe ³²

skin to a heat sink was recently investigated^{32,33}. The concept as generated in this particular investigation applies mainly to mechanically pressurized hard-shell suits, and this type of suit seems to have too many drawbacks to be worth further development²³. Yet, the heat-pipe concept, modified to be compatible with gas-pressurized soft or hard suits, may well prove to be of value as a secondary heat transport system, the function of which is to alleviate the heat loads imposed on the primary system in order to reduce the total power requirement and evaporant expenditure (see Section 4.2).

A schematic representation of a heat-pipe system integrated into a hard-shell suit is given in Figure 2.5. This particular design makes use of two different heat pipe systems arranged in series: a "fixed conductance" flexible heat pipe absorbs heat from the body surface and transmits it to the suit's pressure shell. There, the heat is conducted across the shell to a rigid "variable conductance" type pipe which bypasses the thermal insulation of the suit and transports the heat to the suit's outer shell, where it is rejected either by radiation or by sublimation of ice from the suit's outer surface (see Chapter III of this Report). The heat could also be conducted to the tubing of a liquid loop, which may be located on the inner side of the suit's shell, and conducted to a conventional heat-rejection sink.

The heat pipe scheme is a passive heat transport system with no moving parts (except the control valve in the variable conductance pipe); it is relatively insensitive to external contamination and is capable of transporting large quantities of heat across small temperature gradients³².

On the other hand, heat pipes are not free of flaws. The wicking material inside the pipe tends to degrade with time; in addition, a heat pipe requires a perfectly hermetic sealing to inhibit leakage of the pressurizing gas into the pipe (see Figure 2-6).

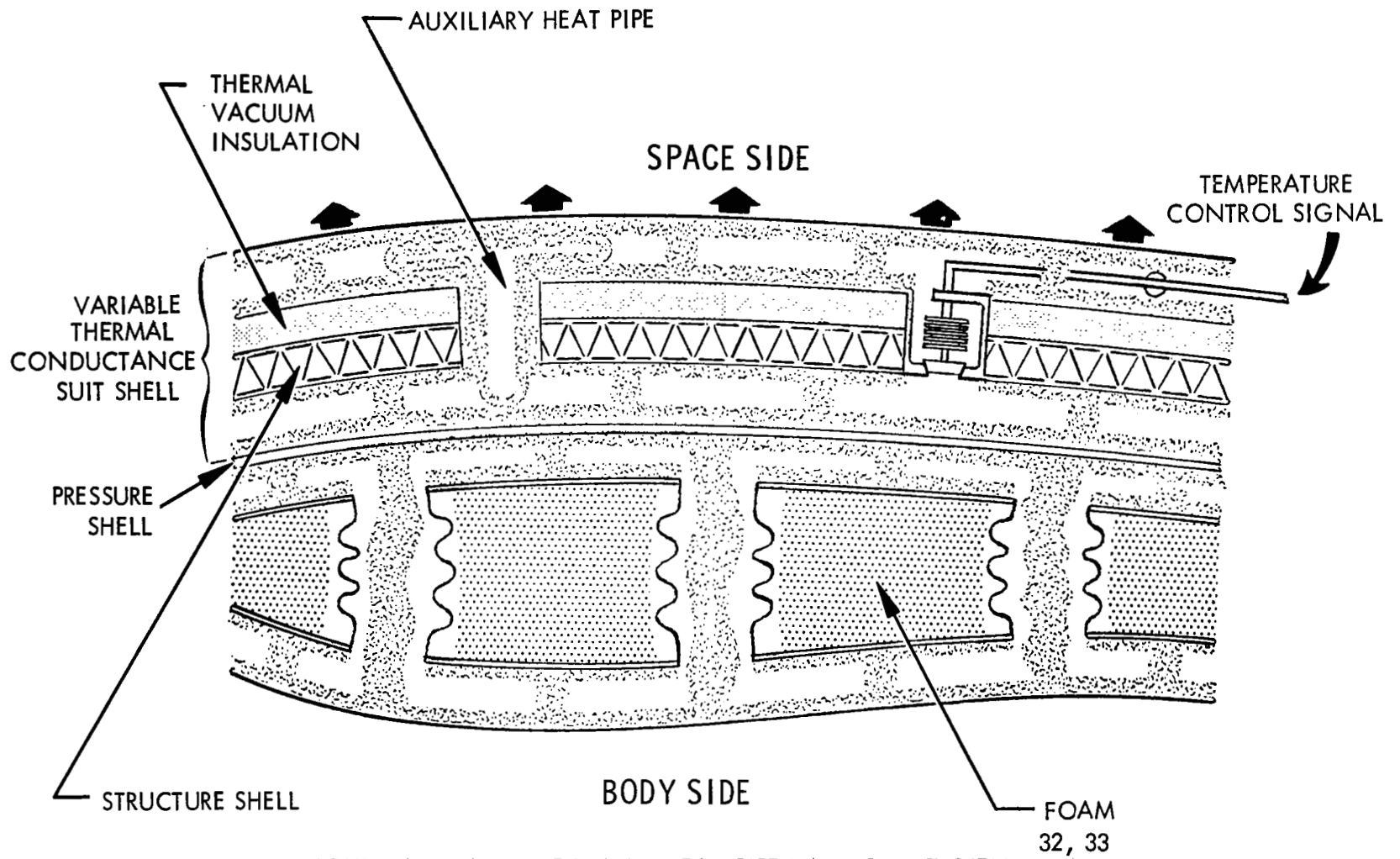


FIGURE 2-5: AN INTEGRATED HEAT-PIPE HARD-SUIT SYSTEM

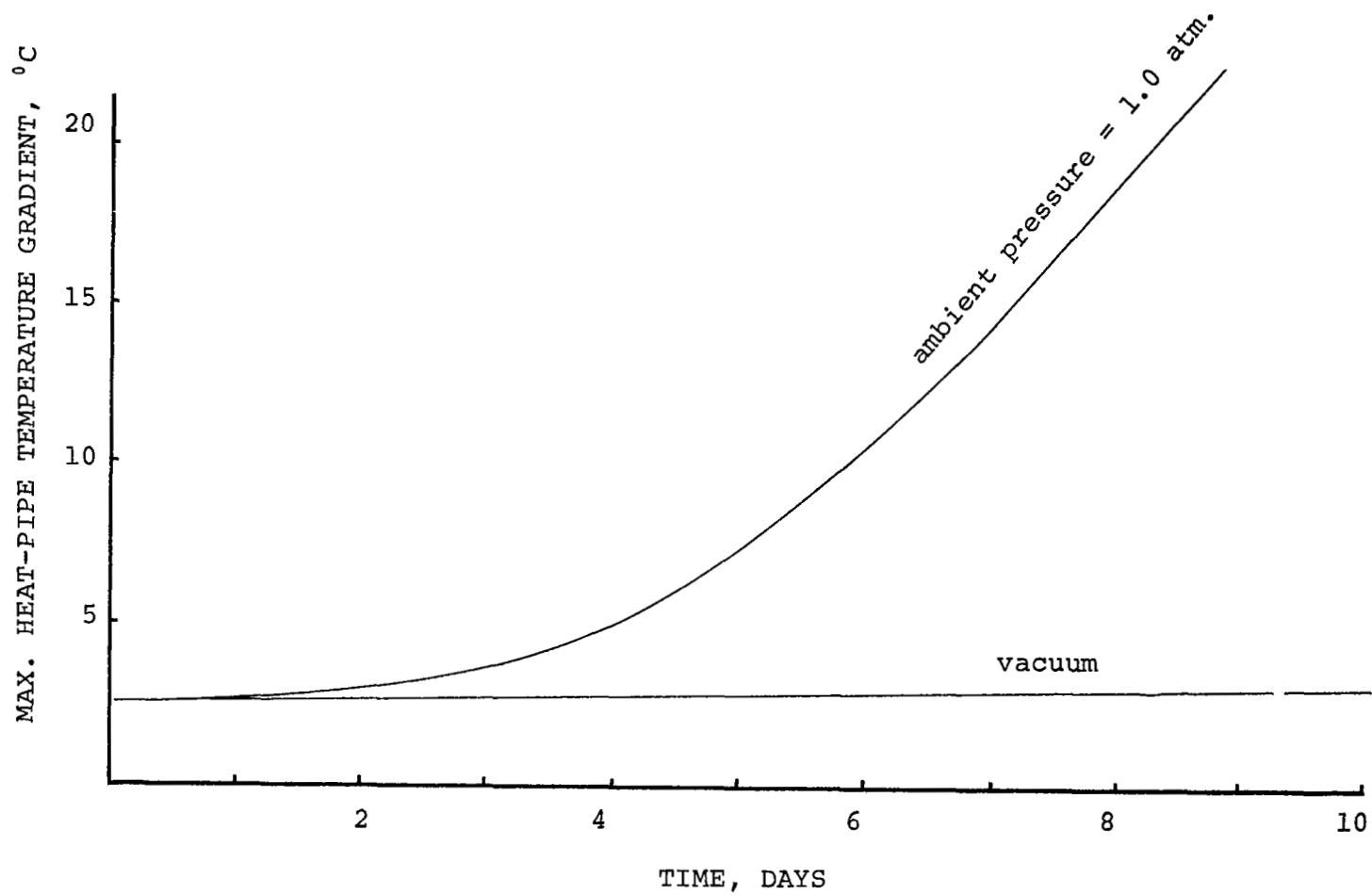


Figure 2-6: Flexible Heat Pipe Performance Deterioration
Due to Leakage. Heat Flux = 5.8 Btu/hr³²

2.4 Recommendations for University Research

- a. Investigate modes of heat-transfer from the body surface to a liquid-cooled garment other than conduction by direct contact. Radiative heat transfer, for example, while not as efficient thermally, may prove to have some advantages from the standpoint of comfort, lack of constraints on movement, ease of donning and doffing and total weight.
- b. Develop an automatic regulator to control LCG cooling rate and astronaut's skin temperature, to avoid excessive sweating or the onset of shivering.
- c. Initiate work on modified, flexible, variable heat pipes compatible with gas-pressurized soft, hybrid and hard space suits.

III. HEAT REJECTION SYSTEMS

A number of portable heat rejection sinks, compatible with space hardware and the space environment, have been proposed over the past few years. Some are true heat rejection devices, rejecting excess heat either by radiation or by dumping high enthalpy expendable overboard; others are in fact heat storage sinks, storing heat during EVA and releasing it during the regeneration cycle on board the spacecraft. Most are compatible with liquid-loop heat transport systems and are described in this chapter. The schemes which are not are discussed in Chapter IV of this report.

At present, techniques which depend on expendables for rejecting heat jettison saturated vapor into space, the enthalpy of which vapor is very nearly the latent heat of vaporization. It has been shown²¹, however, that the total heat rejected could be expressed as

$$Q = \int_{\tau} M \cdot dE - \int_{\tau} (h_0 - E) \cdot dM \quad (3-1)$$

where M = mass of expendable remaining in the system
at any time τ

E = internal energy of the expendable in the system

h_0 = specific enthalpy of escaping vapor

Assuming that $dE = 0$ (the expendable fluid is stored as a liquid at essentially a constant temperature), and recalling that $dM < 0$, it is obvious that to maximize Q , h_0 should be maximized. The latent heat of vaporization, however, is a weak function of the temperature: increasing the temperature of the escaping vapor will increase its enthalpy by only an insignificant amount. On the other hand, h_0 could be increased significantly by accelerating the vapor through a

supersonic nozzle ²¹. While this scheme may add to the complexity of the heat rejection system, careful parametric and feasibility studies may prove to be well worth the effort.

3.1 Ice Sublimators

The vacuum of space offers an attractive method of heat rejection: at ambient pressures below 0.089 psia (the triple point of water), ice sublimates at temperatures below 32⁰ F, affecting the net rejection to space of 1073 Btu/lb (if the feedwater is stored on board in the liquid phase). This is the principle behind the operation of various types of ice sublimators.

Porous Plate Sublimator ^{19,21}

This is the heat rejection system used on the Apollo lunar missions. Water from a pressurized reservoir is fed to one side of a porous metal plate, the other side of which is exposed to space. Water vapor passes through the porous medium and freezes on the outer surface of the plate, thus preventing further water loss under low heat-load conditions. The warm coolant from the astronaut's LCG flows through cored passages in the porous plate and gives up heat to the feedwater; the heat is transferred by conduction into the ice and increases the rate of sublimation, until equilibrium occurs. The unit is therefore said to be self-regulating. A schematic representation of a porous-plate sublimator is shown in Figure 3-1.

Control of LCG coolant inlet temperature can be achieved by either one of the following two methods: in the Apollo PLSS a three-position diverter valve is incorporated in the sublimator inlet (see Figure 3-1), causing some of the warm

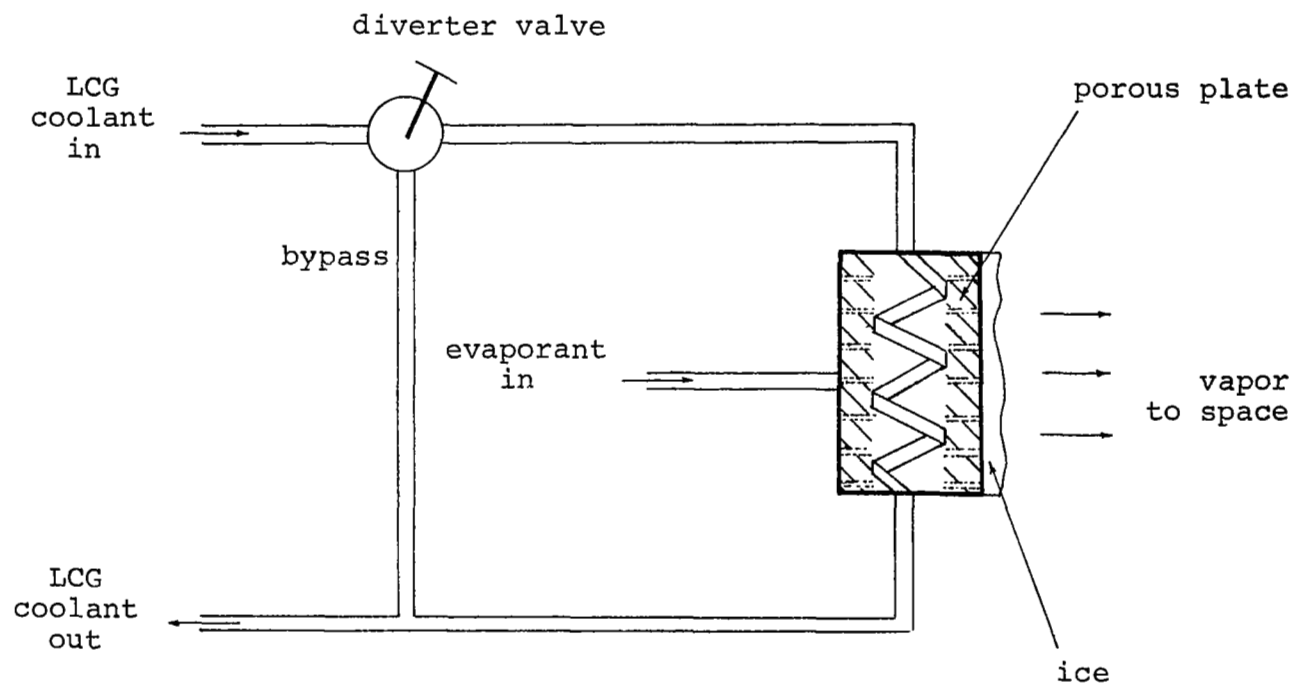


Figure 3-1: Porous-Plate Sublimator

coolant to bypass the heat exchanger, then mix with the cold coolant at the sublimator outlet. This method has the serious disadvantage that at low heat loads, which result in low coolant flow rates through the sublimator, the LCG coolant may freeze in the heat exchanger portion of the sublimator.

To overcome this difficulty the "zone cooling" approach has been proposed¹². Under this scheme, the sublimator surface is divided into zones (four, at the present stage of design), separately fed by water from the evaporant reservoir. The full amount of LCG coolant flows through the heat exchanger at all times. Control is achieved by selecting individual zones through a remote control unit, thus increasing or decreasing the sublimator's active surface area as the need arises. An additional advantage that this method has over the bypass technique is that only a maximum flux of approximately 3 lb/hr of feedwater is controlled, compared with 240 lb/hr of LCG coolant flow. This results in considerably smaller control mechanism and connecting lines.

Ice sublimators have the advantage of using water as the expendable sublimant, water being inexpensive, non-toxic, noncombustible and having high heat of vaporization. Also, ice sublimators are self-regulating and this type of hardware has been space-qualified.

The disadvantages of ice sublimators are that the sublimant, water, is expended to space at the rate of about 12 lbs. for a 4-hours mission; the porous plates are sensitive to contamination; the performance of the porous plates, as that of most capillary devices, may deteriorate with time; under low heat-loads LCG coolant may freeze in the sublimator; and, if the feedwater reservoir is tied into the LCG coolant loop, aeration by the pressurizing gas in the feedwater reservoir may result in cavitation and loss of efficiency in the circulation pump.

It may be worth mentioning at this point that a very significant increase in sublimator efficiency and heat-rejection capability could be achieved by storing the feedwater in the form of subcooled ice²¹ (see Sections 3.4 & 3.5).

Screen Sublimator¹⁹

To overcome the problem of deterioration in the performance of porous plates with time the screen sublimator has been proposed: the porous plate is replaced by a fine-mesh screen and a gelling additive is added to the feedwater. The nature and concentration of the gelling agent is such that the resulting gel can be retained on the screen surface, under operating conditions, by the surface tension of the gelled water interacting with the mesh of the screen.

The gelled feedwater is fed from a pressurized reservoir to the heat exchanger and, under low heat load, forms a frozen layer of ice on the screen surface which is exposed to space. Heat inputs from the circulating LCG coolant will cause the ice to sublime, thus rejecting to space the latent heat of vaporization of the gelled water (which is slightly lower than that of pure H_2O). The gelling additive will float off the subliming surface in the form of fine powder with the escaping vapor.

The advantage of the screen sublimator over the porous plate model is that there is no degradation of capillaries with time and the device is not as sensitive to contamination.

The screen sublimator shares most of the disadvantages of the porous plate sublimator; in addition, the concept has not yet been verified, hardware has not been developed and potential problems stemming from the use of gelled water in space have not yet been investigated.

3.2 Evaporators

Evaporators, like sublimators, rely on space as an infinite low-pressure sink. However, while the phase change in a sublimator is from solid (ice) to gas, the transition taking place in an evaporator is directly from the liquid phase to vapor. If feedwater is stored initially in liquid form, the net heat rejected to space is, of course, the same in both techniques.

An evaporator is shown schematically in Figure 3-2. It consists of a feedwater reservoir which is vented to space through a back-pressure control valve. Warm coolant from the LCG outlet rejects heat to the evaporant, thereby increasing the latter's enthalpy. Vapor escapes through the control valve, rejecting heat to space, until thermal equilibrium is established. LCG coolant temperature is determined by the vapor pressure (and consequently the boiling temperature) of the evaporant which, in turn, is regulated by the setting of the pressure control valve.

A number of schemes which make use of this simple principle and which have a zero-g operation capability have been suggested^{19,21}. All of them share, naturally, the inherent disadvantage that the evaporant is expended to space. If water is used, about 12 lbs. must be jettisoned in a course of a 4-hours EVA mission.

Wick-Fed Evaporator^{7,19}

The evaporant is fed into the evaporator by capillary action along wicks. In the evaporator the feedwater absorbs heat from the LCG coolant in a liquid-to-liquid heat exchanger and evaporates. The vapor flows into a header and is vented to space through the pressure control valve, resulting in the

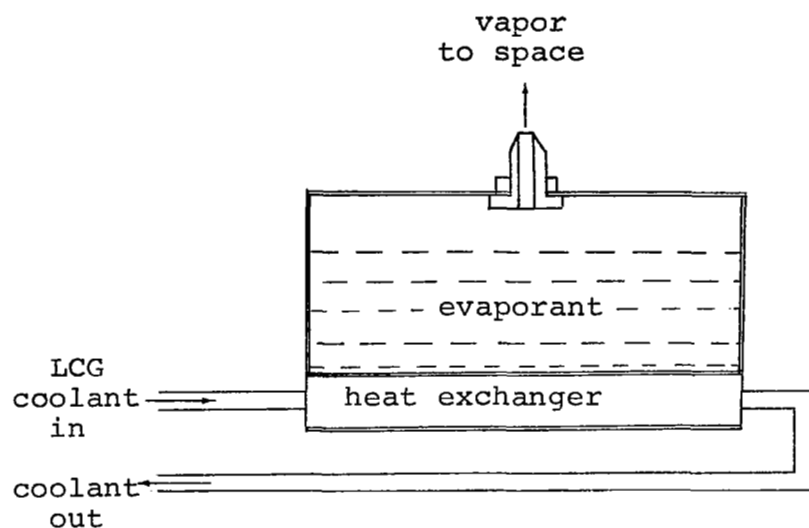


Figure 3-2: Space Evaporator Concept

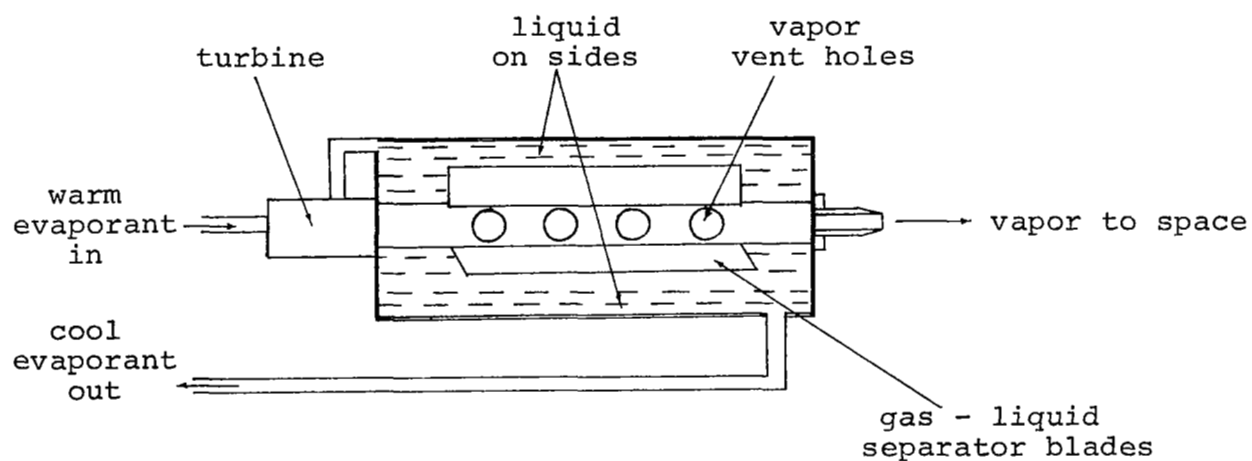


Figure 3-3: Centrifugal Evaporator²¹

rejection to space of 1073 Btu/lb. Water is continuously fed from a gas-pressurized reservoir into the evaporator at the rate required to replenish the evaporated feedwater and keep the wicks wet.

This type of evaporator has been space-qualified and used in space. Its major disadvantage, in addition to the expenditure of water, is that capillary devices are sensitive to external contamination and their performance tends to degrade in time. Also, possible aeration of the evaporant is a problem (see discussion under "Porous Plate Sublimator").

Centrifugal Evaporator²¹

An interesting configuration is that of the centrifugal evaporator (Figure 3-3), in which a separate feedwater system is eliminated and the LCG coolant serves as the evaporant. A centrifugal zero-g gas-liquid separator inside the evaporator is slaved to a water turbine. The turbine, in turn, is driven by the warm coolant from the LCG. The evaporator is vented to space through a back-pressure valve and the temperature of the coolant at the evaporator outlet is regulated by the setting of the valve. The coolant loop must include, naturally, a reservoir for accumulator effect and to replenish the liquid which has been evaporated to space.

In addition to the absence of any capillary devices which may deteriorate with time, this scheme has another advantage: chemical removal of gas contaminants from the suit, especially CO_2 and H_2O vapor, could be eliminated. Instead, gas from the suit is mixed with the cold coolant at the evaporator outlet, then separated in a small gas-liquid separator and ducted back to the suit. The suit gas is thus dehumidified, its H_2O vapor pressure being equal to that of the cold coolant, and CO_2 is removed by dissolution in the water, later to be ejected to space at the low-pressure evaporator. Chemical additives to increase the CO_2 capacity of the coolant may be needed, though.

The disadvantages of this scheme include the expenditure of 12 lbs. of water to space for a 4-hour mission; the turbine and gas separator may add to the complexity and to the power requirements of the system; precautions must be taken to prevent cavitation at the circulation pump by the dissolved gases; and the concept has not yet been verified or tested.

Forced-Vortex Fed Evaporator 7,19

Another attempt to eliminate capillaries and their sensitivity to contamination is represented by the forced-vortex evaporator. Evaporant (water) from a pressurized reservoir is fed into a number of small - diameter metal tubes, which pass through a larger container filled with LCG coolant. The evaporant is forced against the inner wall of the tube by the centrifugal forces caused by a spirally-twisted metal ribbon which is tightly fitted into each tube. The feedwater absorbs the heat load from the LCG coolant and evaporates. The vapor flows into a header and is ejected to space through a back-pressure control valve.

The advantage of this scheme over the wick-fed evaporator is that the heat-transfer surfaces are wetted without dependence on capillary devices. Its disadvantages are that water is expended to space at the rate of 12 lbs. for a 4-hour mission, a distributor is required to equalize the flow of evaporant among the tubes, aeration of the evaporant supply is a possibility (see discussion under "Porous Plate Sublimator") and although performance of a conceptual prototype has been demonstrated no space-qualified hardware has yet been developed.

3.3 Radiators

The performance equation of a radiator is given by

$$\dot{Q} = F A \epsilon \sigma (T_r^4 - T_0^4) \quad (3-2)$$

where \dot{Q} = heat rejection rate, Btu/hr

F = shape (or view) factor

A = radiating surface area, ft²

ϵ = emissivity

σ = Stefan-Boltzmann constant, 0.1714×10^{-8}
(Btu/hr·ft²·°R⁴)

T_r = radiator temperature, °R

T_0 = sink temperature (taken as 0°R for space)

It is not hard to see that an ideal radiator, rejecting the average metabolic heat of 2000 Btu/hr at 530°R (70°F), will have surface area of at least 25 ft². This is clearly too large to mount on an EVA astronaut. Considerable reduction in size can be achieved, however, if the radiator is made to operate at higher temperatures, at the cost of increased power requirements. Table 3-1 shows this trade-off for an ideal radiator, rejecting 2000 Btu/hr.

TABLE 3-1:

RADIATOR TEMPERATURE, AREA AND POWER REQUIREMENT ²¹

T, °F	T, °R	Minimum Power, Watts	Minimum Area, ft ²
70	530	-	A ≈ 25
140	600	76	0.77 A
212	672	160	0.51 A
400	860	360	0.23 A

It should be emphasized that the values given in Table 3-1 are based on the performance of an ideal radiator: irreversible effects in the system may result in considerably larger values of power input and required surface area. In addition, heat inputs from the sun ($440 \text{ Btu/hr}\cdot\text{ft}^2$), the earth, the lunar surface and the spacecraft itself have not been taken into account. A radiator will require complex control system to shield it from these inputs and keep it facing deep space at all times.

Although these difficulties seem to exclude radiators as potential portable heat-rejection devices for EVA, several investigations show that with certain limitations radiators could be used in at least two different types of EVA missions. An integrated system in which the exterior of the space suit serves as the radiating surface might be used in earth orbit²² (see Section 4.2), and radiating plates mounted on a cart could be practical for use during a lunar EVA mission¹⁹.

Lunar-Cart Radiator¹⁹

A radiator mounted on wheels for lunar operation violates Restriction 5 (page 11). However, the system requires no expendables and therefore may be superior to state-of-the-art systems when long EVA periods are involved (see Figs. 5-1 & 5-2).

A schematic representation of the lunar cart is given in Figure 3-4. The cart consists of five plates, one of which is vertical and serves as the radiating panel and four which shield it from heat inputs from the sun and from the lunar surface. All plates are mounted on hinges and can be aligned manually by the astronaut to yield optimum radiation and shielding characteristics. The astronaut's LCG system is coupled with the cart's heat-rejection system by means of an umbilical which provides the astronaut with maneuvering capability in the vicinity of the cart. The astronaut must, of course, pull the cart when changing work stations.

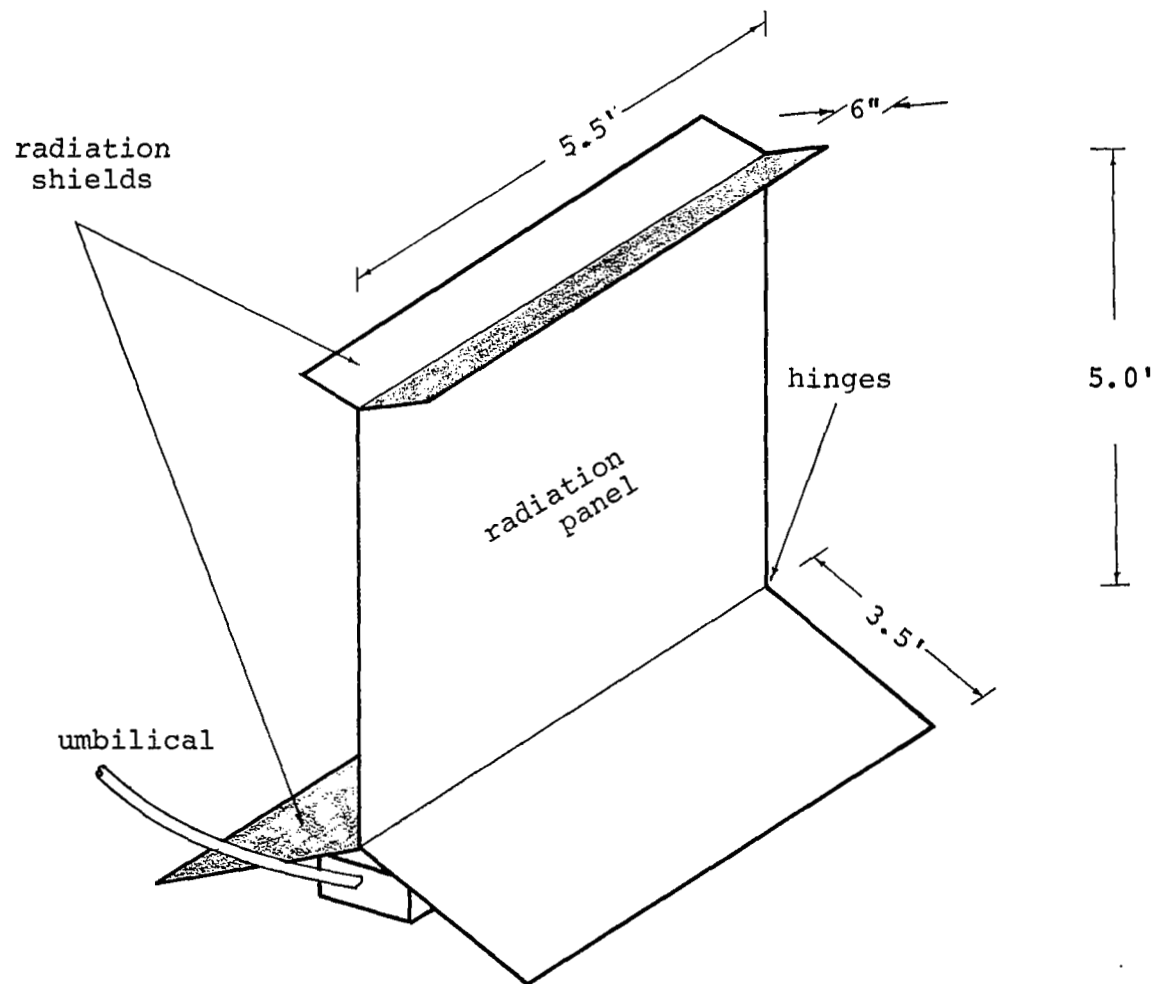


Figure 3-4: Lunar-Cart Radiator¹⁹ (Wheels, Handle etc. Not Shown)

The advantage of the system is that no expendables are needed regardless of the number of EVA hours.

The disadvantages are the large size and mass of the system which may complicate its storage aboard the spacecraft, the astronaut must maneuver the cart and keep it properly oriented as he moves around, the meteoroid penetration hazard, the umbilical connection must be made and broken in hard vacuum and the umbilical may be a source of considerable heat leakage.

3.4 Heat Storage Sinks

On missions involving multiple EVA totalling many hours the weight and volume of the required evaporant rule out the use of evaporators or sublimators as practical methods of heat rejection (see Figures 5-1 & 5-2). On such missions it may be wise to pay a penalty in the form of a more cumbersome heat sink which could be regenerated onboard the spacecraft between EVA for repeated use.

Fusion of Thermal Mass -- Ice ¹⁹

The warm coolant from the LCG outlet is pumped through a bed of ice at 32°F, rejecting 143 Btu to convert 1.0 lb. of ice to liquid H₂O at the same temperature. To facilitate heat transfer from the LCG coolant the ice is stored in a large number of small, spherical capsules made of thin plastic material which is impermeable to water. A piece of heat-conductive wire is placed in each capsule to minimize the quantity of unusable ice. The molten ice is refrozen between EVA onboard the craft and its encapsulation permits repeated use-and-regeneration cycles without deterioration.

The advantages of this method are that it is simple, the heat sink may be used for multiple EVA, no expendables are required and the fusion material, H₂O, is nontoxic, noncombustible, non-irritant and has a high heat of fusion relative to other materials.

The disadvantages are that an ice bed has large weight (approximately 150 lbs. for 4-hour EVA) and volume ($\approx 3\text{ft.}^3$) due to low heat of fusion, 143 Btu/lb., compared to the heat of vaporization, 1073 Btu/lb.; the heat sink must be maintained onboard between missions at temperatures below 32°F, and additives are needed in the LCG coolant (if it is water) to lower its freezing point below 32°F, to keep it from freezing in the heat-exchanger in operation and during the regeneration cycle onboard the spacecraft.

An interesting variation on the theme of ice as a heat sink is the concept of subcooled ice²¹. The temperature dependency of the internal energy of ice with respect to water at 32°F ($E=0$) is given in Table 3-2.

TABLE 3-2:
INTERNAL ENERGY OF ICE

Temperature, °F	E_{ice} , Btu/lb
32	- 143
- 20	- 168
- 40	- 170
- 320	- 310

Thus, storing ice initially at -320°F rather than 32°F may result in the increase of the sink's heat capacity of more than 115%. Of course, a heat sink at that temperature introduces some new problems. The LCG coolant (water) cannot reject its heat directly to the ice mass. A secondary coolant loop is needed, carrying low-freezing-point liquid such as silicone oil. Cooling the ice to cryogenic nitrogen temperature during the regeneration cycle onboard the spacecraft also presents a problem and may increase the complexity of the system.

As noted in Section 3.1, the efficiency of ice sublimators could be increased by more than 30% if the sublimant is stored initially as ice at LN_2 temperature rather than as a liquid. Further studies are needed, however, to determine the feasibility of such scheme as well as possible configurations.

Fusion of Thermal Mass -- Other Fusible Materials

To overcome the need for special measures to prevent LCG coolant freeze-up during the regeneration cycle, some other materials can be used as the fusible mass instead of ice. Some potentially suitable materials are listed in Table 3-3 below.

TABLE 3-3:

POSSIBLE HEAT - SINK FUSIBLE MATERIALS³¹

Material	Density, lb/ft^3	Freezing Point, $^{\circ}\text{F}$	Heat of Fusion, Btu/lb
Ice, H_2O	56	32	143
Tetradecane, $\text{C}_{14}\text{H}_{30}$	48.0	41	98
Hexadecane, $\text{C}_{16}\text{H}_{34}$	48.3	61	100
Octadecane, $\text{C}_{18}\text{H}_{38}$	48.3	81	104

If any of these hydrocarbons is used the heat sink could be regenerated onboard by chilling the LCG coolant (water), and there is no need for additives to lower the coolant's freezing point. Another fringe advantage is that in operation the LCG coolant temperature cannot drop low enough to cause frostbite (39°F). On the other hand, LCG coolant temperature must be at least several degrees higher than the freezing point of the fusible material to transfer heat into the sink. This limits the

sensible heat-transfer capacity of the coolant loop (and probably rules out the use of octadecane as the fusible mass, too). Also, the weight and volume of the hydrocarbon required for one 4-hour EVA are larger than the corresponding values for ice, due to the lower heat of fusion and density of these chemicals. For example, if tetradecane is used, 220 lbs. (at about 4 ft.³) are required for a 4-hour EVA.

3.5 Recommendation for University Research

- a. Investigate the possibilities of improving the performance and lifetime of capillary devices, such as wicks and porous plates, through surface treatment.
- b. Develop ice sublimators which do not require capillary devices, such as porous plates, for operation.
- c. Investigate the concept of the gelled-water screen sublimator and identify problems which may arise from the use of gelled water in the thermal, radiative and gravitational environments of space.
- d. Investigate methods and techniques to optimize the efficiency of expendable refrigerant heat-rejection systems through maximization of the energy of the escaping gas.
- e. Investigate the feasibility of storing sublimants and evaporants in the form of subcooled ice, thus increasing the energy rejection rate per pound of expendable.
- f. Investigate the feasibility of using subcooled ice in regenerable heat sinks.

IV. INTEGRATED COOLING SYSTEMS

Some proposed thermal-control systems do not follow the "classical" configuration of separate heat-transport and heat-sink subsystems. Instead, these two are integrated into a single system which performs both functions. Because of the absence of a separate heat-transport loop the integrated cooling system must be anthropomorphic in order to collect heat from all parts of the astronaut's body. This eliminates the need for a backpack with its extra bulk and weight at the cost of increased complexity of the spacesuit system and, consequently, possible reduction in ease of donning and doffing.

4.1 Liquid Phase Change ^{4,5}

The liquid-phase-change cooling garment (ECGS for evaporative cooling garment system) employs the same method of heat rejection as the evaporator (Section 3.2) and, indeed, can be viewed as an evaporator which depends on direct conduction from the skin, rather than on convection by a liquid loop, as a means of heat transport. A schematic representation of a liquid phase-change garment is given in Figure 4-1.

The garment consists of three layers:

1. A thin, non-permeable, thermally-conductive membrane next to the astronaut's skin.
2. A wicking layer.
3. A flexible interconnecting void which allows gas to flow freely through the structure and which serves as a low-pressure boiler.

The void is vented to space through a regulating valve. The evaporant, water, is fed to the wicking material from a

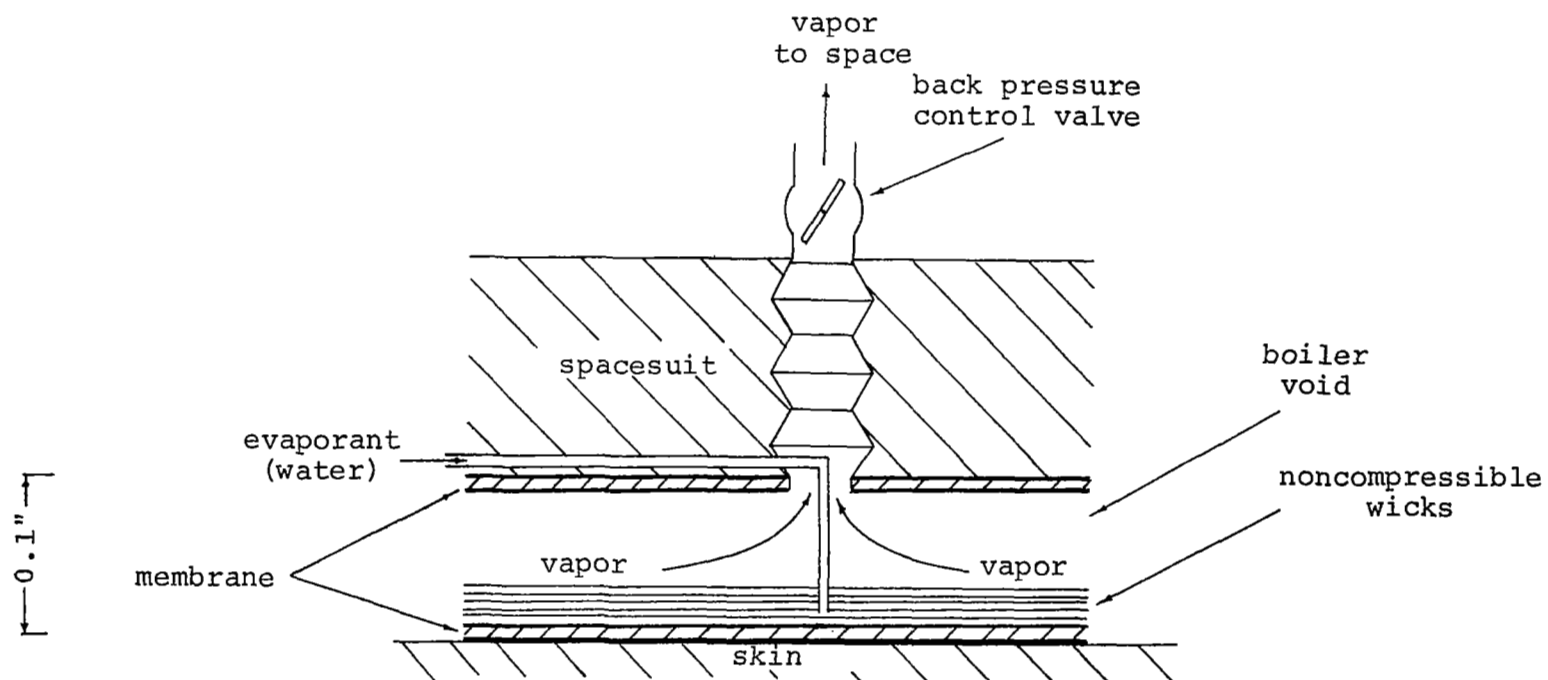


Figure 4-1: Liquid-Phase-Change Cooling Garment (Cross-Section)

reservoir. Metabolic heat is transferred to the moistened wicks by conduction through the membrane. The feedwater evaporates due to the low pressure in the wicking and void structures and the vapor is jettisoned to space, rejecting 1073 Btu of metabolic heat per pound.

At present, the garment system consists of 12 cooling segments. All vacuum lines are ported to a single common outlet to minimize the number of penetrations through the pressure suit and feedwater lines are routed inside the semi-rigid vacuum lines. The separate cooling segments are attached to a next-to-the-skin cotton mesh undergarment, which absorbs possible perspiration, and an outer liner covers all segment penetrations. This results in a smooth outer surface and easy entry into the spacesuit. The articulation areas such as the hips, knees and elbows are not cooled. These areas connect the cooling segments for maximum flexibility and wearing comfort.

The advantages of the liquid-phase-change integrated suit are that it is simple, its heat-rejection rate of more than 5000 Btu/hr can be easily controlled by the back-pressure valve, it uses water as the evaporant, selective temperature control can be obtained by providing each cooling segment with a regulating valve, the system requires no power input if the valve, which is the only moving part, is adjusted manually and a prototype has been built and its performance has been demonstrated.

The disadvantages of this scheme are that the evaporant is expended to space at the rate of 12 lbs. for a 4-hour EVA, the vacuum lines and evaporant-distribution system may constrain astronaut mobility¹⁹, the wicking material may degrade with time as a result of contamination and under low heat loads the astronaut's skin temperature may be reduced to an uncomfortable level¹⁹.

4.2 Radiation

For all practical purposes deep space can be viewed as a black body at 0°R . This raises the possibility of rejecting excess heat to this infinite sink by thermal radiation. The external surface area of a suited astronaut is approximately 25 sq.ft. which is of the order of magnitude of the required radiator area (see Section 3.3). This suggests the possible use of the external surface of the spacesuit as a thermal radiator²².

The astronaut is also subjected to radiant heat inputs from thermal radiation sources such as the sun, the earth, the lunar surface (in the case of a lunar EVA) and the nearby spacecraft. Protection from these inputs can be achieved by shielding or by coating the external surface of the spacesuit with materials which combine low absorptance (α) with low absorptance-to-emittance ratio (α/ϵ).

As noted in Section 1.1 the radiant environment changes considerably in time and space and, consequently, the requirements imposed on the radiating spacesuit surface vary accordingly. As an example, in a 300 n.m. earth-orbit EVA mission²², for $\epsilon=.85$, $\alpha/\epsilon = .2$ and internal suit-wall temperature of 75°F , the maximum possible heat-rejection rate is 2300 Btu/hr in noon orbit and less than 2000 Btu/hr in a twilight orbit ($\alpha/\epsilon > .2$ is the current state-of-the-art in surface coating materials).

Heat transfer from the skin to the radiating surface is best accomplished by direct conduction through the spacesuit, which also provides a means of controlling the heat-rejection rate by varying the thermal conductance of the spacesuit material. This could be accomplished either by pumping helium between the evacuated insulation layers²² or by using variable conductance heat-pipes³³ (see Section 2.3).

The advantages of this heat rejection method are that it requires no expendables and no power input. Its disadvantage is that, under high external heat-loads, it is incapable of rejecting very high metabolic heat rates. It, therefore, can be used only as an auxiliary system, the function of which is to alleviate the heat-loads imposed on the primary system and thus to reduce the total power and expendable requirements. Also, the system's performance may degrade with time due to contamination of the external surface of the suit (and a consequent increase in the value of α/ϵ). A prototype has not yet been developed.

4.3 Phase Change Regenerable Heat-Sink³¹

This technique employs the same concept as the heat storage sink discussed in Section 3.4 except that the fusible mass is stored in close proximity to the skin and heat transfer is accomplished by conduction.

The fusible material is distributed in sealed pockets over the suit's inner surface. The pockets are in direct contact with the skin and metabolic heat is transferred to the mass by conduction through the pocket's outer lining. The sink is cooled during the regeneration cycle onboard the spacecraft, which results in the solidification of the mass.

A serious problem associated with this technique is the formation of a liquid layer of molten heat-sink material between the liquid-solid interface and the skin (shown in Figure 4-2). This layer, which has low thermal conductance, increases the temperature gradient between the skin and the constant-temperature liquid-solid interface and is a limiting factor on the system's operation time.

The problem could be overcome by one of three methods:

1. Use ice, with its high heat of fusion, as the fusible material. Because of the low melting point of water an insulation layer is needed between the skin and the heat sink. By using variable heat-pipes the added insulation of the thickening liquid layer could be compensated for by an increased conductance of the insulation layer.

2. Use a fusible material with a melting point close to the desired skin temperature (see Table 3-3) to minimize the insulation requirements, and improve the thermal conductivity through the liquid layer. This could be accomplished by developing materials with inherently high thermal conductance or by use of thermally-conductive fins extending from the container wall into the fusible mass.

3. Remove the liquid layer as it forms. One way to achieve this is shown schematically in Figure 4-3. A pressurized gas is used to force the solid-liquid interface toward the skin, thus squeezing the molten material into refill bladders. Regeneration of the system will require a warm soak of the suit to melt the fusible material and the application of an external gas pressure at the refill ports to force the liquified material back into the center compartment where it will be cooled and solidified.

The advantages of the integrated heat-sink are that no expendables are needed and that the system can be regenerated onboard the spacecraft for reuse. The disadvantages are that heat transport through a molten liquid layer presents a problem, the system is incapable of absorbing instantaneous metabolic rates in excess of 850-1100 Btu/hr, the hardware may constrain the astronaut's mobility and the design does not account for changes in the astronaut's skin temperature as required for comfort (see Figure 1-2). This is due to the fact that heat absorption by phase-change is a constant temperature process. An additional disadvantage is that no working prototype has yet been developed.

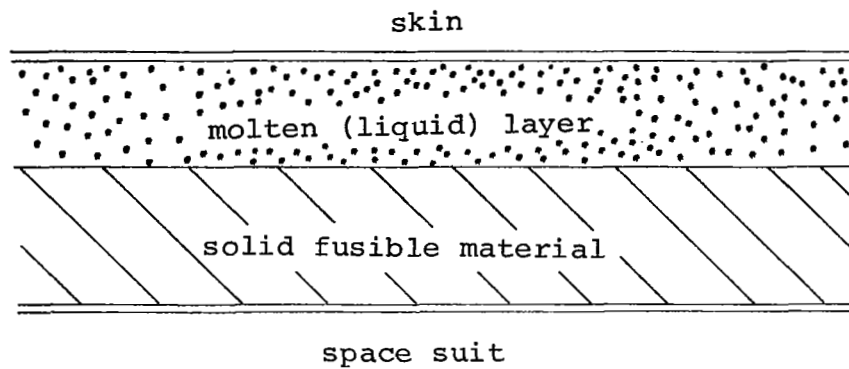


Figure 4-2: Formation of a Liquid Layer

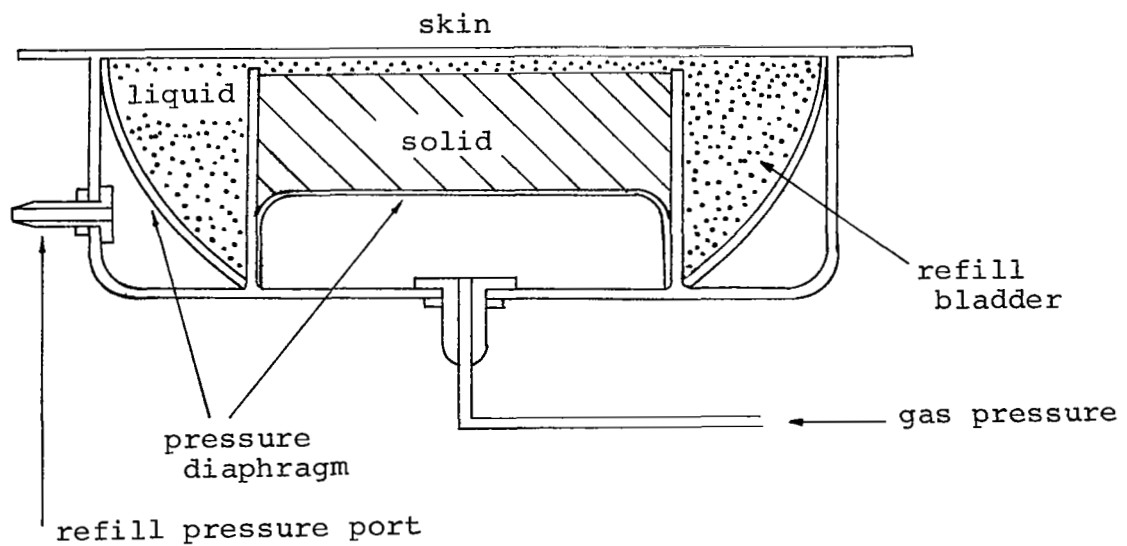


Figure 4-3: Removal of Liquid Layer By Pressurized Gas³¹

4.4 Recommendation for University Research

- a. Investigate the possibility of improving the performance and lifetime of wicking materials.
- b. Initiate the development of an automatic temperature controller compatible with the liquid-phase-change cooling garment.
- c. Investigate the feasibility of an integrated spacesuit radiator as an auxiliary heat-rejection system.

V. SUMMARY

An attempt was made in this report to bring together, in concise form, the most promising concepts in the area of thermal control in portable life-support systems. As is so often the case, the optimal system for any given mission profile probably combines a number of these concepts, exploiting their respective advantages while minimizing the disadvantages. Understandably, the nature of the EVA mission is an important factor in the process of optimizing the thermal control system. Figures 5-1 & 5-2 present the relative advantages of several systems from the standpoint of launch volume and launch weight when the mission consists of repeated 4-hour EVAs. Other factors such as system cost, cost of development, reliability, safety and comfort, to name a few, must also be considered.

With this in mind, the topics which appear to hold most interest for further research are reiterated below:

- a. Better understanding of human thermal processes and heat balance in the space environment is needed.
- b. Develop comfort criteria applicable to EVA under various gravity conditions and different heat-transfer modes.
- c. Develop an automatic regulator to control the cooling rate and the skin temperature for the liquid-cooled garment (LCG) and the integrated liquid-phase-change cooling garment (ECGS).
- d. Develop reliable, flexible, variable-conductance heat-pipes.

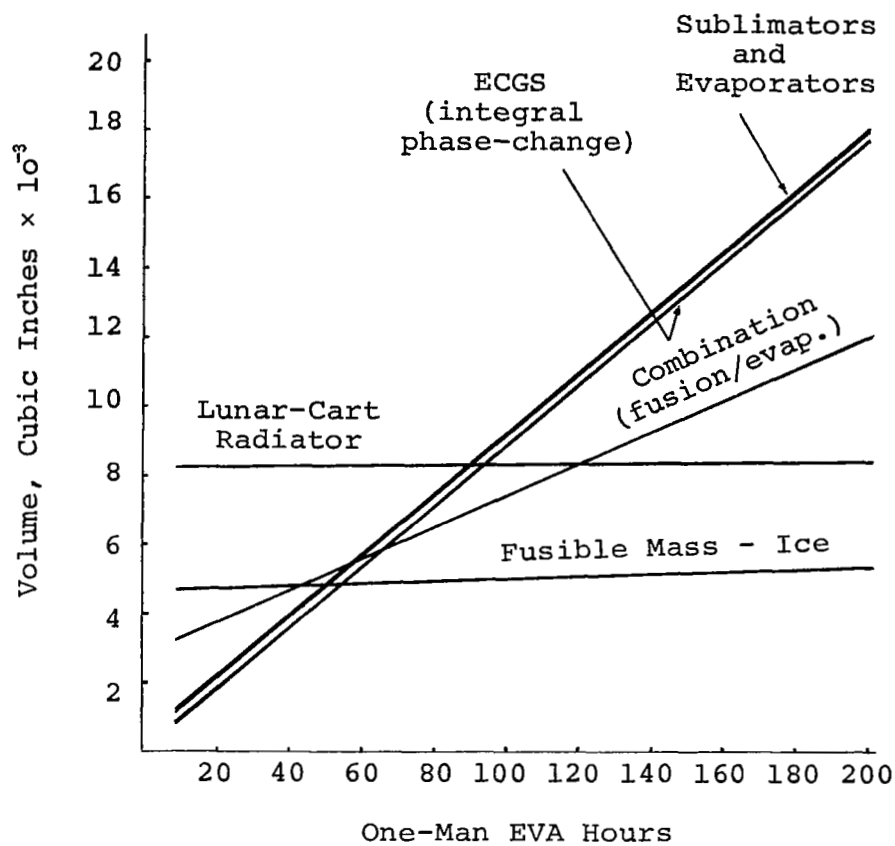


Figure 5-1: Total Launch Volume of Heat-Rejection Sink¹⁹

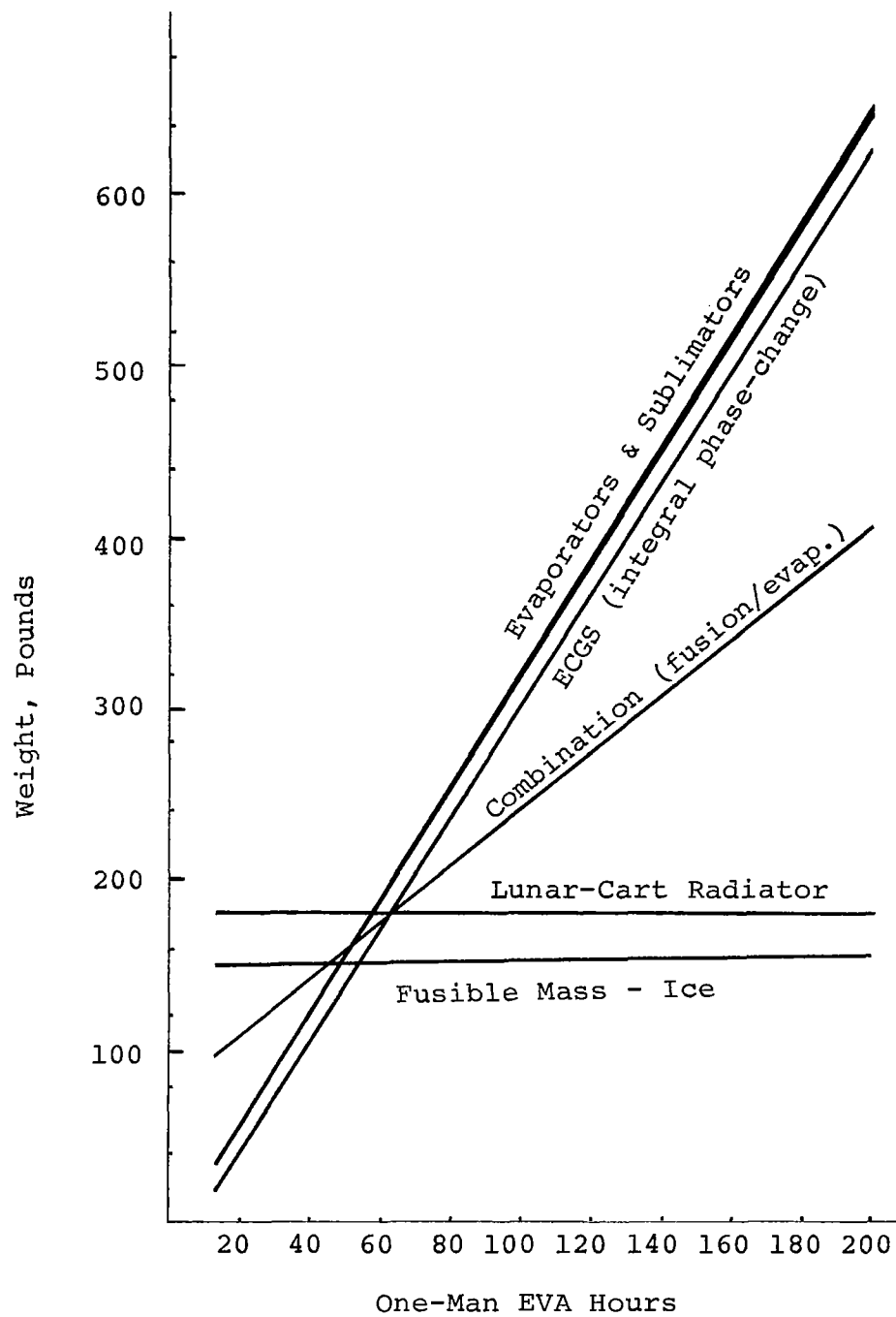


Figure 5-2: Total Launch Weight of Heat-Rejection Sink¹⁹

- e. Improve the performance and the lifetime of capillary devices such as porous plates and wicks.
- f. Maximize the efficiency of expendable-refrigerant systems through the minimization of the energy of the stored refrigerant and the maximization of the energy of the ejected gas.
- g. Investigate the feasibility of auxiliary heat-rejection systems to minimize power and expendable requirements.

ACKNOWLEDGEMENT

This report attempts to bring together in concise form the results of the work of many investigators in the area of thermal control in portable life-support systems. Their great contributions are gratefully acknowledged. In addition, special thanks are extended to Mr. David Richardson of A.D. Little, Inc., to Dr. Ralph Goldman of the U.S. Army Lab in Natick, and to Mr. William Smith of the NASA Office of Life Sciences who reviewed this document and added many critical comments to it.

The project co-ordinator, Mr. John Hatfield, was very helpful during the preparation of the report. This project was supervised by Prof. Laurence Young of the M.I.T. Man-Vehicle Laboratory under NASA Grant NGR-22-009-312.

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